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(दूसरा पुनरीक्षण)

Indian Standard

ROLLING BEARINGS — DYNAMIC LOAD RATINGS AND RATING LIFE

(Second Revision)

ICS 21.100.20

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BUREAU OF INDIAN STANDARDS
MANAK BHAVAN, 9 BAHADUR SHAH ZAFAR MARG
NEW DELHI 110002

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Rolling Bearings Sectional Committee, BP 32

NATIONAL FOREWORD

This Indian Standard (Second Revision) which is identical with ISO 281:1990 'Rolling bearings—Dynamic load ratings and rating life' with Amendment No. 1 and 2 issued by International Organization for Standardization (ISO) was adopted by the Bureau of Indian Standards on the recommendation of the Rolling Bearings Sectional Committee and approval of the Basic and Production Engineering Division Council.

This standard was first published in 1966 and was revised and brought out in four parts in 1983 based on ISO 287-1:1977. This revision is now being taken up to align the standard with the current version of this ISO Standard.

The text of the ISO Standard has been approved as suitable for publication as Indian Standard without deviations. Certain conventions are, however, not identical to those used in Indian Standards. Attention is particularly drawn to the following:

a) Wherever the words 'International Standard' appear referring to this standard, they should be read as 'Indian Standard'.

b) Comma (,) has been used as a decimal marker in the International Standard while in Indian Standards, the current practice is to use a point (.) as the decimal marker.

Amendments No. 1 and 2 to the above International Standard have been printed at the end.

In this adopted standard, reference appears to certain International Standards for which Indian Standards also exist. The corresponding Indian Standards which are to be substituted in their places are listed below along with their degree of equivalence for the editions indicated:

International Standard	*Corresponding Indian Standard*	*Degree of Equivalence*
ISO 76:1987 Rolling bearings — Static load ratings	IS 3823:1988 Rolling bearings — Static load ratings (*second revision*)	Identical
ISO 5593:1997 Rolling bearings — Vocabulary	IS 2399:2002 Rolling bearings — Vocabulary (*second revision*)	do

For the purpose of deciding whether a particular requirement of this standard is complied with, the final value, observed or calculated, expressing the result of a test or analysis, shall be rounded off in accordance with IS 2 : 1960 'Rules for rounding off numerical values (*revised*)'. The number of significant places retained in the rounded off value should be the same as that of the specified value in this standard.

Indian Standard

ROLLING BEARINGS — DYNAMIC LOAD RATINGS AND RATING LIFE

(Second Revision)

1 Scope

This International Standard specifies methods of calculating the basic dynamic load rating of rolling bearings within the size ranges shown in the relevant ISO publications, manufactured from contemporary, commonly used, good quality hardened steel in accordance with good manufacturing practice and basically of conventional design as regards the shape of rolling contact surfaces.

This International Standard also specifies methods of calculating the basic rating life, which is the life associated with 90 % reliability, with commonly used material and manufacturing quality, and with conventional operating conditions. In addition, it specifies methods of calculating adjusted rating life, in which various reliabilities, special bearing properties and specific operating conditions are taken into account by means of life adjustment factors.

This International Standard is not applicable to designs where the rolling elements operate directly on a shaft or housing surface, unless that surface is equivalent in all respects to the bearing ring (or washer) raceway it replaces.

Double row radial bearings and double direction thrust bearings are, when referred to in this International Standard, presumed to be symmetrical.

Further limitations concerning particular types of bearings are included in the relevant clauses.

2 Normative references

The following standards contain provisions which, through reference in this text, constitute provisions of this International Standard. At the time of publication, the editions indicated were valid. All standards are subject to revision, and parties to agreements based on this International Standard are encouraged to investigate the possibility of applying the most recent editions of the standards indicated below. Members of IEC and ISO maintain registers of currently valid International Standards.

ISO 76 : 1987, *Rolling bearings — Static load ratings.*

ISO 5593 : 1984, *Rolling bearings — Vocabulary.*

3 Definitions

For the purposes of this International Standard, the definitions given in ISO 5593, together with the following, apply.

3.1 life: For an individual rolling bearing, the number of revolutions which one of the bearing rings (or washers) makes in relation to the other ring (or washer) before the first evidence of fatigue develops in the material of one of the rings (or washers) or rolling elements.

3.2 reliability (in the context of bearing life): For a group of apparently identical rolling bearings, operating under the same conditions, the percentage of the group that is expected to attain or exceed a specified life.

The reliability of an individual rolling bearing is the probability that the bearing will attain or exceed a specified life.

3.3 basic rating life: For an individual rolling bearing, or a group of apparently identical rolling bearings operating under the same conditions, the life associated with 90 % reliability, with contemporary, commonly used material and manufacturing quality, and under conventional operating conditions.

3.4 adjusted rating life: The rating life obtained by adjustment of the basic rating life for a desired reliability level, special bearing properties and specific operating conditions.

3.5 basic dynamic radial load rating: That constant stationary radial load which a rolling bearing could theoretically endure for a basic rating life of one million revolutions.

In the case of a single row angular contact bearing, the radial load rating refers to the radial component of that load which causes a purely radial displacement of the bearing rings in relation to each other.

3.6 basic dynamic axial load rating: That constant centric axial load which a rolling bearing could theoretically endure for a basic rating life of one million revolutions.

3.7 dynamic equivalent radial load: That constant stationary radial load under the influence of which a rolling bearing would have the same life as it will attain under the actual load conditions.

3.8 dynamic equivalent axial load: That constant centric axial load under the influence of which a rolling bearing would have the same life as it will attain under the actual load conditions.

3.9 roller diameter applicable in the calculation of load ratings: The diameter at the middle of the roller.

NOTE – For a tapered roller this is equal to the mean value of the diameters at the theoretically sharp corners at the large end and at the small end of the roller.

For an asymmetrical convex roller this is an approximation of the diameter at the point of contact between the roller and the ribless raceway at zero load.

3.10 roller length applicable in the calculation of load ratings: The theoretical maximum length of contact between a roller and that raceway where the contact is shortest.

NOTE – This is normally taken to be either the distance between the theoretically sharp corners of the roller minus the roller chamfers or the raceway width excluding the grinding undercuts, whichever is the smaller.

3.11 nominal contact angle: The angle between a plane perpendicular to the bearing axis and the nominal line of action of the resultant of the forces transmitted by a bearing ring to a rolling element.

3.12 pitch diameter of a ball set: The diameter of the circle containing the centres of the balls in one row in a bearing.

3.13 pitch diameter of a roller set: The diameter of the circle intersecting the roller axes at the middle of the rollers in one row in a bearing.

3.14 conventional operating conditions: Conditions which may be assumed to prevail for a bearing which is properly mounted and protected from foreign matter, adequately lubricated, conventionally loaded, not exposed to extreme temperature and not run at exceptionally low or high speed.

4 Symbols

C_r = basic dynamic radial load rating, in newtons

C_a = basic dynamic axial load rating, in newtons

C_{0r} = basic static radial load rating[1], in newtons

C_{0a} = basic static axial load rating[1], in newtons

D_w = ball diameter, in millimetres

D_{we} = roller diameter applicable in the calculation of load ratings, in millimetres

D_{pw} = pitch diameter of ball or roller set, in millimetres

F_r = bearing radial load = radial component of actual bearing load, in newtons

F_a = bearing axial load = axial component of actual bearing load, in newtons

L_{10} = basic rating life, in million revolutions

L_{na} = adjusted rating life, in million revolutions

L_{we} = roller length applicable in the calculation of load ratings, in millimetres

P_r = dynamic equivalent radial load, in newtons

P_a = dynamic equivalent axial load, in newtons

X = dynamic radial load factor

Y = dynamic axial load factor

Z = number of balls or rollers in a single row bearing; number per row of a multi-row bearing with the same number of balls or rollers per row

a_1 = life adjustment factor for reliability (see 9.3)

a_2 = life adjustment factor for special bearing properties (see 9.4)

a_3 = life adjustment factor for operating conditions (see 9.5)

b_m = rating factor for contemporary, normally used material and manufacturing quality, the value of which varies with bearing type and design

e = limiting value of F_a/F_r for the applicability of different values of factors X and Y

f_c = factor which depends on the geometry of the bearing components, the accuracy to which the various components are made, and the material

f_0 = factor which depends on the geometry of the bearing components and on the applicable stress level[1]

i = number of rows of balls or rollers in a bearing

α = nominal contact angle of a bearing, in degrees

5 Radial ball bearings

5.1 Basic dynamic radial load rating

The basic dynamic radial load rating, C_r, for radial and angular contact ball bearings is given by

$$C_r = b_m f_c (i \cos\alpha)^{0,7} Z^{2/3} D_w^{1,8}$$

for $D_w < 25,4$ mm

$$C_r = 3,647\, b_m f_c (i \cos\alpha)^{0,7} Z^{2/3} D_w^{1,4}$$

for $D_w > 25,4$ mm

Values of b_m are given in table 1. Values of f_c are given in table 2. They apply to bearings with a cross-sectional raceway groove radius not larger than 0,52 D_w in radial and angular contact groove ball bearing inner rings and 0,53 D_w in radial and angular contact groove ball bearing outer rings and self-aligning ball bearing inner rings.

The load-carrying ability of a bearing is not necessarily increased by the use of a smaller groove radius, but it is reduced by the use of larger radii than those indicated above.

---

[1] For definition, calculation method and values, see ISO 76.

Table 1 — Values of b_m for radial ball bearings

Bearing type	b_m
Radial and angular contact groove ball bearings (except filling slot and insert bearings) and self-aligning ball bearings	1,3
Filling slot bearings	1,1
Insert bearings	1

5.1.1 Bearing combinations

5.1.1.1 When calculating the basic radial load rating for two similar single row radial contact groove ball bearings mounted side-by-side on the same shaft such that they operate as a unit (paired mounting), the pair is considered as one double row radial contact bearing.

5.1.1.2 When calculating the basic radial load rating for two similar single row angular contact ball bearings mounted side-by-side on the same shaft such that they operate as a unit (paired mounting), in "back-to-back" or "face-to face" arrangement, the pair is considered as one double row angular contact bearing.

5.1.1.3 The basic radial load rating for two or more similar single row angular contact ball bearings mounted side-by-side on the same shaft such that they operate as a unit (paired or stack mounting) in "tandem" arrangement, properly manufactured and mounted for equal load distribution, is the number of bearings to the power of 0,7, times the rating of one single row bearing.

5.1.1.4 If, for some technical reason, the bearing arrangement is regarded as a number of single row bearings which are replaceable independently of each other, then 5.1.1.3 does not apply.

5.2 Dynamic equivalent radial load

The dynamic equivalent radial load, P_r, for radial and angular contact ball bearings, under constant radial and axial loads, is given by

$$P_r = XF_r + YF_a$$

Values of X and Y are given in table 3.

5.2.1 Bearing combinations

5.2.1.1 When calculating the equivalent radial load for two similar single row angular contact ball bearings mounted side-by-side on the same shaft such that they operate as a unit (paired mounting) in "back-to-back" or "face-to-face" arrangement, the pair is considered as one double row angular contact bearing.

5.2.1.2 When calculating the equivalent radial load for two or more similar single row ball bearings mounted side-by-side on the same shaft such that they operate as a unit (paired or stack mounting) in "tandem" arrangement, the values of X and Y for a single row bearing are used. The "relative axial load" (see table 3) is established by using $i = 1$ and the F_a and C_{0r} values which both refer to one of the bearings only (even though the F_r and F_a values referring to the total loads are used for the calculation of the equivalent load for the complete arrangement).

5.3 Basic rating life

5.3.1 The basic rating life, L_{10}, for a radial ball bearing is given by

$$L_{10} = \left(\frac{C_r}{P_r}\right)^3$$

The values of C_r and P_r are calculated in accordance with 5.1 and 5.2.

This life formula is also used for the evaluation of the life of two or more single row bearings operating as a unit, as referred to in 5.1.1. In this case, the load rating C_r is calculated for the complete bearing arrangement and the equivalent load P_r is calculated for the total loads acting on the arrangement, using the values of X and Y indicated in 5.2.1.2.

5.3.2 The life formula gives satisfactory results for a broad range of bearing loads. However, extra-heavy loads may cause detrimental plastic deformations at the ball/raceway contacts. The user should therefore consult the bearing manufacturer to establish the applicability of the life formula in cases where P_r exceeds C_{0r} or 0,5 C_r, whichever is the smaller.

Table 2 — Values of f_c for radial ball bearings

$\frac{D_w \cos\alpha}{D_{pw}}$ [1]	f_c			
	Single row radial contact groove ball bearings and single and double row angular contact groove ball bearings	**Double row radial contact groove ball bearings**	**Single row and double row self-aligning ball bearings**	**Single row radial contact separable ball bearings (magneto bearings)**
0,01	29,1	27,5	9,9	9,4
0,02	35,8	33,9	12,4	11,7
0,03	40,3	38,2	14,3	13,4
0,04	43,8	41,5	15,9	14,9
0,05	46,7	44,2	17,3	16,2
0,06	49,1	46,5	18,6	17,4
0,07	51,1	48,4	19,9	18,5
0,08	52,8	50	21,1	19,5
0,09	54,3	51,4	22,3	20,6
0,1	55,5	52,6	23,4	21,5
0,11	56,6	53,6	24,5	22,5
0,12	57,5	54,5	25,6	23,4
0,13	58,2	55,2	26,6	24,4
0,14	58,8	55,7	27,7	25,3
0,15	59,3	56,1	28,7	26,2
0,16	59,6	56,5	29,7	27,1
0,17	59,8	56,7	30,7	27,9
0,18	59,9	56,8	31,7	28,8
0,19	60	56,8	32,6	29,7
0,2	59,9	56,8	33,5	30,5
0,21	59,8	56,6	34,4	31,3
0,22	59,6	56,5	35,2	32,1
0,23	59,3	56,2	36,1	32,9
0,24	59	55,9	36,8	33,7
0,25	58,6	55,5	37,5	34,5
0,26	58,2	55,1	38,2	35,2
0,27	57,7	54,6	38,8	35,9
0,28	57,1	54,1	39,4	36,6
0,29	56,6	53,6	39,9	37,2
0,3	56	53	40,3	37,8
0,31	55,3	52,4	40,6	38,4
0,32	54,6	51,8	40,9	38,9
0,33	53,9	51,1	41,1	39,4
0,34	53,2	50,4	41,2	39,8
0,35	52,4	49,7	41,3	40,1
0,36	51,7	48,9	41,3	40,4
0,37	50,9	48,2	41,2	40,7
0,38	50	47,4	41	40,8
0,39	49,2	46,6	40,7	40,9
0,4	48,4	45,8	40,4	40,9

[1] Values of f_c for intermediate values of $\frac{D_w \cos\alpha}{D_{pw}}$ are obtained by linear interpolation.

Table 3 — Values of X and Y for radial ball bearings

Bearing type		"Relative axial load" 1), 2)		Single row bearings				Double row bearings				e
				$\frac{F_a}{F_r} < e$		$\frac{F_a}{F_r} > e$		$\frac{F_a}{F_r} < e$		$\frac{F_a}{F_r} > e$		
				X	Y	X	Y	X	Y	X	Y	
Radial contact groove ball bearings		$\frac{f_0 F_a}{C_{0r}}$ 3)	$\frac{F_a}{iZD_w^2}$									
		0,172	0,172	1	0	0,56	2,3	1	0	0,56	2,3	0,19
		0,345	0,345				1,99				1,99	0,22
		0,689	0,689				1,71				1,71	0,26
		1,03	1,03				1,55				1,55	0,28
		1,38	1,38				1,45				1,45	0,3
		2,07	2,07				1,31				1,31	0,34
		3,45	3,45				1,15				1,15	0,38
		5,17	5,17				1,04				1,04	0,42
		6,89	6,89				1				1	0,44
Angular contact groove ball bearings		$\frac{f_0 iF_a}{C_{0r}}$ 3)	$\frac{F_a}{ZD_w^2}$									
	$\alpha = 5°$	0,173	0,172	1	0	For this type use the X, Y and e values applicable to single row radial contact groove ball bearings		1	2,78	0,78	3,74	0,23
		0,346	0,345						2,4		3,23	0,26
		0,692	0,689						2,07		2,78	0,3
		1,04	1,03						1,87		2,52	0,34
		1,38	1,38						1,75		2,36	0,36
		2,08	2,07						1,58		2,13	0,4
		3,46	3,45						1,39		1,87	0,45
		5,19	5,17						1,26		1,69	0,5
		6,92	6,89						1,21		1,63	0,52
	$\alpha = 10°$	0,175	0,172	1	0	0,46	1,88	1	2,18	0,75	3,06	0,29
		0,35	0,345				1,71		1,98		2,78	0,32
		0,7	0,689				1,52		1,76		2,47	0,36
		1,05	1,03				1,41		1,63		2,29	0,38
		1,4	1,38				1,34		1,55		2,18	0,4
		2,1	2,07				1,23		1,42		2	0,44
		3,50	3,45				1,1		1,27		1,79	0,49
		5,25	5,17				1,01		1,17		1,64	0,54
		7	6,89				1		1,16		1,63	0,54
	$\alpha = 15°$	0,178	0,172	1	0	0,44	1,47	1	1,65	0,72	2,39	0,38
		0,357	0,345				1,4		1,57		2,28	0,4
		0,714	0,689				1,3		1,46		2,11	0,43
		1,07	1,03				1,23		1,38		2	0,46
		1,43	1,38				1,19		1,34		1,93	0,47
		2,14	2,07				1,12		1,26		1,82	0,5
		3,57	3,45				1,02		1,14		1,66	0,55
		5,35	5,17				1		1,12		1,63	0,56
		7,14	6,89				1		1,12		1,63	0,56
	$\alpha = 20°$	—	—	1	0	0,43	1	1	1,09	0,7	1,63	0,57
	$\alpha = 25°$	—	—			0,41	0,87		0,92	0,67	1,41	0,68
	$\alpha = 30°$	—	—			0,39	0,76		0,78	0,63	1,24	0,8
	$\alpha = 35°$	—	—			0,37	0,66		0,66	0,6	1,07	0,95
	$\alpha = 40°$	—	—			0,35	0,57		0,55	0,57	0,93	1,14
	$\alpha = 45°$	—	—			0,33	0,5		0,47	0,54	0,81	1,34
Self-aligning ball bearings				1	0	0,4	$0{,}4 \cot\alpha$	1	$0{,}42 \cot\alpha$	0,65	$0{,}65 \cot\alpha$	$1{,}5 \tan\alpha$
Single row radial contact separable ball bearings (magneto bearings)				1	0	0,5	2,5	—	—	—	—	0,2

1) Permissible maximum value depends on the bearing design (internal clearance and raceway groove depth). Use the first or second column depending on available information.

2) Values of X, Y and e for intermediate "relative axial loads" and/or contact angles are obtained by linear interpolation.

3) For values of f_0, see ISO 76.

6 Thrust ball bearings

6.1 Basic dynamic axial load rating

6.1.1 Single row bearings

The basic dynamic axial load rating, C_a, for single row, single or double direction thrust ball bearings is given by

$$C_a = b_m f_c Z^{2/3} D_w^{1,8}$$

for $D_w < 25{,}4$ mm and $\alpha = 90°$

$$C_a = b_m f_c (\cos\alpha)^{0,7} \tan\alpha\, Z^{2/3} D_w^{1,8}$$

for $D_w < 25{,}4$ mm and $\alpha \neq 90°$

$$C_a = 3{,}647\, b_m f_c Z^{2/3} D_w^{1,4}$$

for $D_w > 25{,}4$ mm and $\alpha = 90°$

$$C_a = 3{,}647\, b_m f_c (\cos\alpha)^{0,7} \tan\alpha\, Z^{2/3} D_w^{1,4}$$

for $D_w > 25{,}4$ mm and $\alpha \neq 90°$

where

Z is the number of balls carrying load in one direction;

$b_m = 1{,}3$.

Values of f_c are given in table 4 and apply to bearings with a cross-sectional raceway groove radius not larger than $0{,}54\ D_w$.

The load-carrying ability of a bearing is not necessarily increased by the use of a smaller groove radius, but is reduced by the use of a larger groove radius than that indicated above.

6.1.2 Bearings with two or more rows of balls

The basic dynamic axial load rating, C_a, for thrust ball bearings, with two or more rows of similar balls carrying load in the same direction, is given by

$$C_a = (Z_1 + Z_2 + \ldots + Z_n) \times \left[\left(\frac{Z_1}{C_{a1}}\right)^{10/3} + \left(\frac{Z_2}{C_{a2}}\right)^{10/3} + \ldots + \left(\frac{Z_n}{C_{an}}\right)^{10/3}\right]^{-3/10}$$

The load ratings C_{a1}, C_{a2}, . . . , C_{an} for the rows with Z_1, Z_2, . . . , Z_n balls are calculated from the appropriate single row bearing formula given in 6.1.1.

Table 4 — Values of f_c for thrust ball bearings

$\frac{D_w}{D_{pw}}$ [1]	f_c	$\frac{D_w \cos\alpha}{D_{pw}}$ [1]	f_c		
	$\alpha = 90°$		$\alpha = 45°$ [2]	$\alpha = 60°$	$\alpha = 75°$
0,01	36,7	0,01	42,1	39,2	37,3
0,02	45,2	0,02	51,7	48,1	45,9
0,03	51,1	0,03	58,2	54,2	51,7
0,04	55,7	0,04	63,3	58,9	56,1
0,05	59,5	0,05	67,3	62,6	59,7
0,06	62,9	0,06	70,7	65,8	62,7
0,07	65,8	0,07	73,5	68,4	65,2
0,08	68,5	0,08	75,9	70,7	67,3
0,09	71	0,09	78	72,6	69,2
0,1	73,3	0,1	79,7	74,2	70,7
0,11	75,4	0,11	81,1	75,5	
0,12	77,4	0,12	82,3	76,6	
0,13	79,3	0,13	83,3	77,5	
0,14	81,1	0,14	84,1	78,3	
0,15	82,7	0,15	84,7	78,8	
0,16	84,4	0,16	85,1	79,2	
0,17	85,9	0,17	85,4	79,5	
0,18	87,4	0,18	85,5	79,6	
0,19	88,8	0,19	85,5	79,6	
0,2	90,2	0,2	85,4	79,5	
0,21	91,5	0,21	85,2		
0,22	92,8	0,22	84,9		
0,23	94,1	0,23	84,5		
0,24	95,3	0,24	84		
0,25	96,4	0,25	83,4		
0,26	97,6	0,26	82,8		
0,27	98,7	0,27	82		
0,28	99,8	0,28	81,3		
0,29	100,8	0,29	80,4		
0,3	101,9	0,3	79,6		
0,31	102,9				
0,32	103,9				
0,33	104,8				
0,34	105,8				
0,35	106,7				

[1] Values of f_c for $\frac{D_w}{D_{pw}}$ or $\frac{D_w \cos\alpha}{D_{pw}}$ and/or contact angles other than those shown in the table are obtained by linear interpolation.

[2] For thrust bearings $\alpha > 45°$. Values for $\alpha = 45°$ are given to permit interpolation of values for α between 45° and 60°.

6.2 Dynamic equivalent axial load

The dynamic equivalent axial load, P_a, for thrust ball bearings with $\alpha \neq 90°$, under constant radial and axial loads, is given by

$$P_a = XF_r + YF_a$$

Values of X and Y are given in table 5.

Thrust ball bearings with $\alpha = 90°$ can support axial loads only. The dynamic equivalent axial load for this type of bearing is given by

$$P_a = F_a$$

6.3 Basic rating life

6.3.1 The basic rating life, L_{10}, for a thrust ball bearing is given by

$$L_{10} = \left(\frac{C_a}{P_a}\right)^3$$

The values of C_a and P_a are calculated in accordance with 6.1 and 6.2.

6.3.2 The life formula gives satisfactory results for a broad range of bearing loads. However, extra-heavy loads may cause detrimental plastic deformations at the ball/raceway contacts.

Table 5 — Values of X and Y for thrust ball bearings

α [1]	Single direction bearings[2]		Double direction bearings				e
	$\frac{F_a}{F_r} > e$		$\frac{F_a}{F_r} < e$		$\frac{F_a}{F_r} > e$		
	X	Y	X	Y	X	Y	
45° [3]	0,66	1	1,18	0,59	0,66	1	1,25
50°	0,73		1,37	0,57	0,73		1,49
55°	0,81		1,6	0,56	0,81		1,79
60°	0,92		1,9	0,55	0,92		2,17
65°	1,06		2,3	0,54	1,06		2,68
70°	1,28		2,9	0,53	1,28		3,43
75°	1,66		3,89	0,52	1,66		4,67
80°	2,43		5,86	0,52	2,43		7,09
85°	4,8		11,75	0,51	4,8		14,29
$\alpha \neq 90°$	$1{,}25\tan\alpha\left(1 - \frac{2}{3}\sin\alpha\right)$	1	$\frac{20}{13}\tan\alpha\left(1 - \frac{1}{3}\sin\alpha\right)$	$\frac{10}{13}\left(1 - \frac{1}{3}\sin\alpha\right)$	$1{,}25\tan\alpha\left(1 - \frac{2}{3}\sin\alpha\right)$	1	$1{,}25\tan\alpha$

1) Values of X, Y and e for intermediate values of α are obtained by linear interpolation.

2) $\frac{F_a}{F_r} < e$ is unsuitable for single direction bearings.

3) For thrust bearings $\alpha > 45°$. Values for $\alpha = 45°$ are given to permit interpolation of values for α between 45° and 50°.

The user should therefore consult the bearing manufacturer to establish the applicability of the life formula in cases where P_a exceeds $0{,}5\ C_a$.

7 Radial roller bearings

7.1 Basic dynamic radial load rating

The basic dynamic radial load rating, C_r, for a radial roller bearing is given by

$$C_r = b_m f_c (i L_{we} \cos\alpha)^{7/9} Z^{3/4} D_{we}^{29/27}$$

Values of b_m are given in table 6. Values of f_c are given in table 7. They are maximum values, only applicable to roller bearings in which, under a bearing load, the material stress is substantially uniform along the most heavily loaded roller/raceway contact.

Smaller values of f_c than those given in table 7 should be used if, under load, an accentuated stress concentration is present in some part of the roller/raceway contact. Such stress concentrations must be expected, for example, at the centre of nominal point contacts, at the extremities of line contacts, in bearings where the rollers are not accurately guided and in bearings with rollers longer than 2,5 times the roller diameter.

Table 6 — Values of b_m for radial roller bearings

Bearing type	b_m
Cylindrical roller bearings, tapered roller bearings and needle roller bearings with machined rings	1,1
Drawn cup needle roller bearings	1
Spherical roller bearings	1,15

Table 7 — Maximum values of f_c for radial roller bearings (see 7.1)

$\frac{D_{we}\cos\alpha}{D_{pw}}$ [1]	f_c
0,01	52,1
0,02	60,8
0,03	66,5
0,04	70,7
0,05	74,1
0,06	76,9
0,07	79,2
0,08	81,2
0,09	82,8
0,1	84,2
0,11	85,4
0,12	86,4
0,13	87,1
0,14	87,7
0,15	88,2
0,16	88,5
0,17	88,7
0,18	88,8
0,19	88,8
0,2	88,7
0,21	88,5
0,22	88,2
0,23	87,9
0,24	87,5
0,25	87
0,26	86,4
0,27	85,8
0,28	85,2
0,29	84,5
0,3	83,8

1) Values of f_c for intermediate values of $\frac{D_{we}\cos\alpha}{D_{pw}}$ are obtained by linear interpolation.

7.1.1 Bearing combinations

7.1.1.1 When calculating the basic radial load rating for two similar single row roller bearings mounted side-by-side on the same shaft such that they operate as a unit (paired mounting) in "back-to-back" or "face-to-face" arrangement, the pair is considered as one double row bearing.

7.1.1.2 If, for some technical reason, the bearing arrangement is regarded as two bearings which are replaceable independently of each other, then 7.1.1.1 does not apply.

7.1.1.3 The basic radial load rating for two or more similar single row roller bearings mounted side-by-side on the same shaft such that they operate as a unit (paired or stack mounting) in "tandem" arrangement, properly manufactured and mounted for equal load distribution, is the number of bearings to the power of 7/9, times the rating of one single row bearing.

7.1.1.4 If, for some technical reason, the bearing arrangement is regarded as a number of single row bearings which are replaceable independently of each other, then 7.1.1.3 does not apply.

7.2 Dynamic equivalent radial load

The dynamic equivalent radial load, P_r, for radial roller bearings with $\alpha \neq 0°$, under constant radial and axial loads, is given by

$$P_r = XF_r + YF_a$$

Values of X and Y are given in table 8.

The dynamic equivalent radial load for radial roller bearings with $\alpha = 0°$, and subjected to radial load only, is given by

$$P_r = F_r$$

NOTE — The ability of radial roller bearings with $\alpha = 0°$ to support axial loads varies considerably with bearing design and execution. The bearing user should therefore consult the bearing manufacturer for recommendations regarding the evaluation of equivalent load and life in cases where bearings with $\alpha = 0°$ are subjected to axial load.

7.2.1 Bearing combinations

7.2.1.1 When calculating the equivalent radial load for two similar single row angular contact roller bearings mounted side-by-side on the same shaft such that they operate as a unit (paired mounting) in "back-to-back" or "face-to-face" arrangement, and which, according to 7.1.1, is considered as one double row bearing, the values of X and Y for double row bearings given in table 8 are used.

7.2.1.2 When calculating the equivalent radial load for two or more similar single row angular contact roller bearings mounted side-by-side on the same shaft such that they operate as a unit (paired or stack mounting) in "tandem" arrangement, the X and Y factors for single row bearings given in table 8 are used.

Table 8 — Values of X and Y for radial roller bearings

Bearing type	$\frac{F_a}{F_r} < e$		$\frac{F_a}{F_r} > e$		e
	X	Y	X	Y	
Single row, $\alpha \neq 0$	1	0	0,4	0,4 $\cot\alpha$	1,5 $\tan\alpha$
Double row, $\alpha \neq 0$	1	0,45 $\cot\alpha$	0,67	0,67 $\cot\alpha$	1,5 $\tan\alpha$

7.3 Basic rating life

7.3.1 The basic rating life, L_{10}, for a radial roller bearing is given by

$$L_{10} = \left(\frac{C_r}{P_r}\right)^{10/3}$$

The values of C_r and P_r are calculated in accordance with 7.1 and 7.2.

This life formula is also used for the evaluation of the life of two or more single row bearings operating as a unit, as referred to in 7.1.1. In this case, the load rating C_r is calculated for the complete bearing arrangement and the equivalent load P_r is calculated for the total loads acting on the arrangement, using the values of X and Y indicated in 7.2.1.

7.3.2 The life formula gives satisfactory results for a broad range of bearing loads. However, extra-heavy load may cause accentuated stress concentrations in some part of the roller/raceway contacts. The user should therefore consult the bearing manufacturer to establish the applicability of the life formula in cases where P_r exceeds 0,5 C_r.

8 Thrust roller bearings

8.1 Basic dynamic axial load rating

8.1.1 Single row bearings

8.1.1.1 A thrust roller bearing is considered as a single row bearing only if all rollers carrying load in the same direction contact the same washer raceway area.

The basic dynamic axial load rating, C_a, for single row, single or double direction thrust roller bearings is given by

$$C_a = b_m f_c L_{we}^{7/9} Z^{3/4} D_{we}^{29/27}$$
$$\text{for } \alpha = 90°$$

$$C_a = b_m f_c (L_{we} \cos\alpha)^{7/9} \tan\alpha \, Z^{3/4} D_{we}^{29/27}$$
$$\text{for } \alpha \neq 90°$$

where Z is the number of rollers carrying load in one direction.

8.1.1.2 If several rollers, on the same side of the bearing axis, are located with their axes coinciding, these rollers are considered as one roller with a length L_{we} equal to the sum of the lengths (see 3.10) of the several rollers.

Values of b_m are given in table 9. Values of f_c are given in table 10. They are maximum values, only applicable to roller bearings in which, under a bearing load, the material stress is substantially uniform along the most heavily loaded roller/raceway contact.

Smaller values of f_c than those given in table 10 should be used if, under load, an accentuated stress concentration is present in some part of the roller/raceway contact. Such stress concentrations must be expected, for example, at the centre of nominal point contacts, at the extremities of line contacts, in bearings where the rollers are not accurately guided and in bearings with rollers longer than 2,5 times the roller diameter.

Smaller values of f_c should also be considered for thrust roller bearings in which the geometry causes excessive slip in the roller/raceway contact areas, for example bearings with cylindrical rollers which are long in relation to the pitch diameter of the roller set.

Table 9 — Values of b_m for thrust roller bearings

Bearing type	b_m
Cylindrical roller bearings and needle roller bearings	1
Tapered roller bearings	1,1
Spherical roller bearings	1,15

8.1.2 Bearings with two or more rows of rollers

The basic dynamic axial load rating, C_a, for thrust roller bearings with two or more rows of rollers carrying load in the same direction is given by

$$C_a = (Z_1 L_{we1} + Z_2 L_{we2} + \dots + Z_n L_{wen}) \times$$

$$\times \left[\left(\frac{Z_1 L_{we1}}{C_{a1}} \right)^{9/2} + \left(\frac{Z_2 L_{we2}}{C_{a2}} \right)^{9/2} + \dots \right.$$

$$\left. \dots + \left(\frac{Z_n L_{wen}}{C_{an}} \right)^{9/2} \right]^{-2/9}$$

The load ratings C_{a1}, C_{a2}, . . . , C_{an} for the rows with Z_1, Z_2, . . . , Z_n rollers of lengths L_{we1}, L_{we2}, . . . , L_{wen} are calculated from the appropriate single row bearing formula given in 8.1.1.1.

Rollers and/or portions of rollers which contact the same washer raceway area belong to one row.

8.1.3 Bearing combinations

8.1.3.1 The basic axial load rating for two or more similar single direction thrust roller bearings mounted side-by-side on the same shaft such that they operate as a unit (paired or stack mounting) in "tandem" arrangement, properly manufactured and mounted for equal load distribution, is the number of bearings to the power of 7/9, times the rating of one bearing.

8.1.3.2 If, for some technical reason, the bearing arrangement is regarded as a number of single direction bearings which are replaceable independently of each other, then 8.1.3.1 does not apply.

Table 10 — Maximum values of f_c for thrust roller bearings (see 8.1.1.2)

$\frac{D_{we}}{D_{pw}}$ 1)	f_c	$\frac{D_{we}\cos\alpha}{D_{pw}}$ 1)	f_c		
	$\alpha = 90°$		$\alpha = 50°$ 2)	$\alpha = 65°$ 3)	$\alpha = 80°$ 4)
0,01	105,4	0,01	109,7	107,1	105,6
0,02	122,9	0,02	127,8	124,7	123
0,03	134,5	0,03	139,5	136,2	134,3
0,04	143,4	0,04	148,3	144,7	142,8
0,05	150,7	0,05	155,2	151,5	149,4
0,06	156,9	0,06	160,9	157	154,9
0,07	162,4	0,07	165,6	161,6	159,4
0,08	167,2	0,08	169,5	165,5	163,2
0,09	171,7	0,09	172,8	168,7	166,4
0,1	175,7	0,1	175,5	171,4	169
0,11	179,5	0,11	177,8	173,6	171,2
0,12	183	0,12	179,7	175,4	173
0,13	186,3	0,13	181,1	176,8	174,4
0,14	189,4	0,14	182,3	177,9	175,5
0,15	192,3	0,15	183,1	178,8	176,3
0,16	195,1	0,16	183,7	179,3	
0,17	197,7	0,17	184	179,6	
0,18	200,3	0,18	184,1	179,7	
0,19	202,7	0,19	184	179,6	
0,2	205	0,2	183,7	179,3	
0,21	207,2	0,21	183,2		
0,22	209,4	0,22	182,6		
0,23	211,5	0,23	181,8		
0,24	213,5	0,24	180,9		
0,25	215,4	0,25	179,8		
0,26	217,3	0,26	178,7		
0,27	219,1				
0,28	220,9				
0,29	222,7				
0,3	224,3				

1) Values of f_c for intermediate values of $\frac{D_{we}}{D_{pw}}$ or $\frac{D_{we}\cos\alpha}{D_{pw}}$ are obtained by linear interpolation.

2) Applicable for $45° < \alpha < 60°$

3) Applicable for $60° \leqslant \alpha < 75°$

4) Applicable for $75° \leqslant \alpha < 90°$

8.2 Dynamic equivalent axial load

The dynamic equivalent axial load, P_a, for thrust roller bearings with $\alpha \neq 90°$, under constant radial and axial loads, is given by

$$P_a = XF_r + YF_a$$

Values of X and Y are given in table 11.

Thrust roller bearings with $\alpha = 90°$ can support axial loads only. The dynamic equivalent axial load for this type of bearing is given by

$$P_a = F_a$$

8.3 Basic rating life

8.3.1 The basic rating life, L_{10}, for a thrust roller bearing is given by

$$L_{10} = \left(\frac{C_a}{P_a} \right)^{10/3}$$

Table 11 — Values of X and Y for thrust roller bearings

Bearing type	$\frac{F_a}{F_r} < e$		$\frac{F_a}{F_r} > e$		e
	X	Y	X	Y	
Single direction, $\alpha \neq 90°$	— 1)	— 1)	$\tan\alpha$	1	$1{,}5 \tan\alpha$
Double direction, $\alpha \neq 90°$	$1{,}5 \tan\alpha$	0,67	$\tan\alpha$	1	$1{,}5 \tan\alpha$

1) $\frac{F_a}{F_r} < e$ is unsuitable for single direction bearings.

The values of C_a and P_a are calculated in accordance with 8.1 and 8.2.

This life formula is also used for the evaluation of the life of two or more single direction thrust roller bearings operating as a unit, as referred to in 8.1.3. In this case, the load rating C_a is calculated for the complete bearing arrangement and the equivalent load P_a is calculated for the total loads acting on the arrangement, using the values of X and Y given for single direction bearings in 8.2.

8.3.2 The life formula gives satisfactory results for a broad range of bearing loads. However, extra-heavy load may cause accentuated stress concentrations in some part of the roller/raceway contacts. The user should therefore consult the bearing manufacturer to establish the applicability of the life formula in cases where P_a exceeds 0,5 C_a.

9 Adjusted rating life

9.1 General

It is often satisfactory to use the basic rating life, L_{10}, as a criterion of bearing performance. This life is associated with 90 % reliability, with contemporary, commonly used material and manufacturing quality, and under conventional operating conditions.

However, for many applications it may be desirable to calculate the life for a different reliability and/or for special bearing properties and operating conditions which deviate from the conventional in such a way that it is justified to take their influence into special consideration.

The adjusted rating life, L_{na}, i.e. the basic rating life adjusted for a reliability of $(100 - n)$ %, for special bearing properties and for specific operating conditions, is given by

$$L_{na} = a_1 a_2 a_3 L_{10}$$

Values of a_1 are given in table 12 (see 9.3). Values of a_2 and a_3 are discussed in 9.4 and 9.5. The value of L_{10} is calculated in accordance with 5.3, 6.3, 7.3 and 8.3.

9.2 Limitations

In addition to the required fatigue life, other factors, such as maximum permissible bearing deflection and minimum shaft and housing strength, should be given due consideration when selecting the size of bearings for a given application. Particular discretion shall be exercised when using adjusted rating life values which are based on values of a_2 and a_3 greater than 1.

9.3 Life adjustment factor for reliability, a_1

Reliability is defined in 3.2. The adjusted rating life is calculated in accordance with 9.1. Values of the life adjustment factor a_1 are given in table 12.

Table 12 — Life adjustment factor for reliability, a_1

Reliability %	L_{na}	a_1
90	L_{10a}	1
95	L_{5a}	0,62
96	L_{4a}	0,53
97	L_{3a}	0,44
98	L_{2a}	0,33
99	L_{1a}	0,21

9.4 Life adjustment factor for special bearing properties, a_2

9.4.1 A bearing may acquire special properties, as regards life, by the use of a special type and quality of material and/or special manufacturing processes and/or special design. Such special life properties are taken into account by the application of the life adjustment factor a_2.

The present state of knowledge does not make it possible to define relationships between the values of a_2 and quantifiable characteristics of the material or bearing raceway geometry, for example. The values of a_2 have therefore to be based on experience, and may usually be obtained from the manufacturer of the bearing.

9.4.2 The use of a certain steel analysis and/or process as such is not sufficient justification for the use of an a_2 value other than 1. Values of a_2 greater than 1 may, however, be applicable to bearings made of steel of particularly low impurity content or of special analysis. However, if a reduced life is expected because of a hardness reduction caused by special heat treatment, this should be considered by the selection of a correspondingly reduced a_2 value.

9.4.3 A special design involving an increased or reduced uniformity of the stress in the contacts between rolling elements and raceways should also be considered in the selection of the value of a_2.

9.4.4 It may not be assumed that the use of a special material, process or design will overcome a deficiency in lubrication. Values of a_2 greater than 1 should therefore normally not be applied if a_3 is less than 1 because of such deficiency.

9.5 Life adjustment factor for operating conditions, a_3

9.5.1 Of the operating conditions directly influencing bearing life, the direction and magnitude of the load are considered in the calculation of the equivalent load (5.2, 6.2, 7.2 and 8.2), and deviations from normal load distribution are discussed in the Introduction.

Operating conditions which remain to be taken into account here include the adequacy of the lubrication (at the operating speed and temperature), presence of foreign matter, and conditions causing changes in material properties (for example, high temperature causing reduced hardness). The influence on bearing life of such conditions may be taken into account by the introduction of a life adjustment factor a_3.

9.5.2 The calculation of basic rating life in this International Standard assumes that the lubrication is adequate, i.e. that the lubricant film in the rolling element/raceway contacts has a thickness which is equal to or slightly greater than the composite roughness of the contact surfaces. Where this requirement is fulfilled, a_3 is equal to 1, provided a lower value does not apply, for example because of a change in material properties caused by the operating conditions.

9.5.3 Values of a_3 less than 1 should be considered, for example, where the kinematic viscosity of the lubricant, at the operating temperature, is less than 13 mm²/s[1] for ball bearings or 20 mm²/s for roller bearings, and/or where the rotational speed is exceptionally low (i.e. the number of revolutions per minute times D_{pw} is less than 10 000).

Values of a_3 greater than 1 may be considered only where the lubrication conditions are so favourable that the probability of failure caused by surface distress is greatly reduced.

Manufacturers of bearings are expected to supply recommendations regarding appropriate values of a_3 to be used in the calculation of adjusted rating life in accordance with 9.1.

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1) 1 $\text{mm}^2/\text{s} = 1$ cSt

Annex A
(informative)

Bibliography

ISO/TR 8646 : 1985, *Explanatory notes on ISO 281/1-1977.*

AMENDMENT 1

Page 1, Scope

To comply with the changed wording for the designation of factor b_m, the first and the second paragraph of clause 1 (Scope) on page 1 of ISO 281 are slightly changed. The new wording is:

This International Standard specifies methods of calculating the basic dynamic load rating of rolling bearings within the size ranges shown in the relevant ISO publications, manufactured from contemporary, commonly used, high quality, hardened bearing steel in accordance with good manufacturing practice and basically of conventional design as regards the shape of rolling contact surfaces.

This International Standard also specifies methods of calculating the basic rating life, which is the life associated with 90 % reliability, with commonly used high quality material, good manufacturing quality, and with conventional operating conditions. In addition, it specifies methods of calculating adjusted rating life, in which various reliabilities, special bearing properties and specific operating conditions are taken into account by means of life adjustment factors.

Page 2, Symbols

Change the wording of the definition of the factor b_m to read:

b_m = rating factor for contemporary, commonly used, high quality, hardened bearing steel in accordance with good manufacturing practices, the value of which varies with bearing type and design.

Page 3, Table 1

Change the value of the factor b_m from 1 to 1,3 for insert bearings.

AMENDMENT 2

1 Symbols

a_{XYZ} is the ife modification factor, based on a systems approach in accordance with 2.4;

a_1 is the life adjustment factor for reliability;

a_2, a_3, a_4, a_5, a_m are the interdependent life adjustment factors for various influences in accordance with 2.3 and 2.4.2;

e is the Weibull exponent;

L_{na} is the adjusted rating life, in millions of revolutions;

L_{nm} is the modified rating life, in millions of revolutions;

L_{10} is the basic rating life, in millions of revolutions;

n is the probability of failure, in %;

S is the reliability (probability of survival) in %, within the range 100 to 0;

κ is the ratio of viscosity = ν / ν_1;

Λ is the film parameter, ratio of film thickness to composite surface roughness;

ν is the actual lubricant viscosity at the operating temperature, in square millimetres per second;

ν_1 is the required viscosity at the operating temperature to obtain adequate lubrication, in square millimetres per second;

σ is the real stress used in fatigue criterion, in megapascals;

σ_u is the endurance stress limit used in fatigue criterion, in megapascals.

2 Modified rating life

2.1 General

It is often sufficient to use the basic rating life, L_{10}, as a criterion of bearing performance. This life is associated with 90 % reliability.

However, for some applications it may be of interest to calculate the life for a higher reliability, and for many applications it is desirable to take into account the influence of the bearing quality and operating conditions in a more accurate and complete way. The modified rating life, L_{nm}, meets this demand. [The index n represents probability of failure in % and $(100 - n)$ the probability of survival (also expressed as the reliability).]

The life L_{nm}, i.e. the basic rating life modified for a reliability of $(100 - n)$ %, for special bearing properties and for specific operating conditions, can be calculated with the formula

$$L_{nm} = a_1 \, a_{XYZ} \, L_{10} \qquad (1)$$

Values of the life adjustment factor a_1 are given in Table 1.

2.2 Life adjustment factor for reliability, a_1

Table 1 — Life adjustment factor for reliability, a_1

Reliability S	L_{nm}	a_1
90	L_{10m}	1
95	L_{5m}	0,62
96	L_{4m}	0,53
97	L_{3m}	0,44
98	L_{2m}	0,33
99	L_{1m}	0,21

Table 1 is based on a Weibull exponent of $e = 1{,}5$. It is possible to calculate a_1 for other exponent values (see annex A).

2.3 Life modification factor, a_{XYZ}

In ISO 281-1:1977 the factors a_2 and a_3 were introduced to consider the influence of material and lubrication conditions on bearing life. However, it has been recognised that a_2 and a_3 are interdependent, and this has led many bearing manufacturers to use a combined a_{23} factor. The scope of this combined factor a_{23} has been extended with the introduction of the a_{XYZ} factor to cover additional interdependent influences.

These interdependencies imply that

$$a_{XYZ} = f(a_2, a_3, a_m) \qquad (2)$$

Modern technology makes it possible to determine a_{XYZ} by combining computer supported theory with empirical tests and practical experience. Besides bearing type, the factor a_{XYZ} can include the influence of:

- material (e.g. cleanliness, hardness, surface structure, fatigue limit, response to temperature);
- lubrication (e.g. viscosity, bearing speed, bearing size, type of lubricant, additives);
- environment (e.g. contamination level, humidity);
- contaminant particles (e.g. hardness, size, form, material);
- internal stresses in the rings (e.g. from the manufacturing process, from inner ring interference after mounting);
- mounting (e.g. handling damages, misalignment);
- bearing load.

2.4 Systems approach to life calculations

2.4.1 Modified life calculation by means of systems approach

Below a certain load a modern high quality bearing can attain an infinite life if the lubrication conditions, the cleanliness and other operating conditions are favourable. The real contact stress when the fatigue load limit is reached is of the order of 1 500 MPa for normal bearing steels.

In many applications however, contact stresses are larger than this value and, in addition, the operating conditions can further reduce the bearing life.

It is possible to relate all influences to the applied stresses and to the strength of the material, e.g.:

— indentations give rise to edge stresses;

— a thin oil film increases the shear stresses in the contact region between raceway and rolling element;

— an increased temperature reduces the fatigue limit of the material, i.e. its strength;

— a tight inner ring fit gives rise to hoop stresses.

The different influences on bearing life are dependent on each other. A systems approach to the fatigue life calculation is therefore appropriate in which the influence on the life of the system due to variation and interaction of interdependent factors will be considered.

Diagrams or equations can be made up expressing a_{XYZ} as a function of σ_u/σ, the endurance stress limit divided by the real stress, with as many influencing factors as possible considered (see Figure 1).

In Figure 1, the diagram for a given lubrication condition also illustrates how a_{XYZ} asymptotically approaches infinity, if the real stress, σ, is decreased down to the endurance stress limit, σ_u, when a fatigue criterion applies. Traditionally, the orthogonal shear stress has been used as a fatigue criterion in bearing life calculations (see reference [1] in the Bibliography). The diagram in Figure 1 can therefore also be based on endurance strength in shear.

Manufacturers are expected to be responsible for the advice given about the calculation of a_{XYZ} as a function of real stresses, endurance stress limit and operating conditions. The letters XYZ in the designation a_{XYZ} indicate that a manufacturer or organization can select any combination and number of letters.

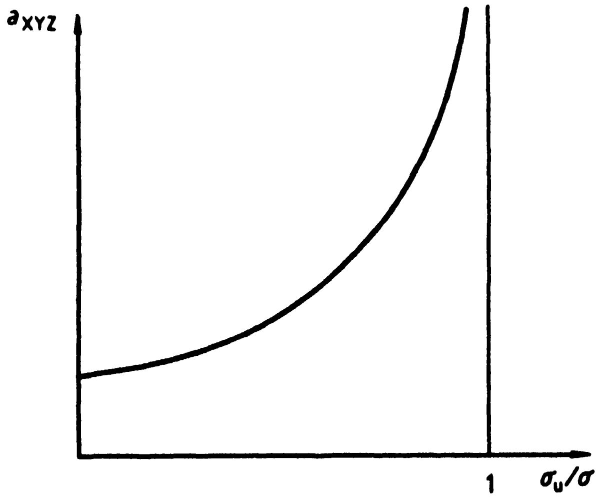

Figure 1 — Life modification factor a_{XYZ}

The lubrication conditions can be expressed either by

— the viscosity ratio $\kappa = \nu / \nu_1$, which is defined as the real, actual oil viscosity ν at the operating temperature divided by the oil viscosity ν_1, required for adequate lubrication,

or by

— the film parameter Λ, which is defined as the ratio of the oil film thickness to the root mean square of the surface irregularities of the contacting surfaces.

The factors κ and Λ are both measures of the risk of intermetallic contact through the lubricant film. Manufacturers are expected to supply recommendations regarding the calculation of the factors.

2.4.2 Modified life calculation by means of multiplication factors

As a special case of the systems approach, it is possible to calculate a_{XYZ} with the aid of multiplication factors, e.g.

$$a_{\text{XYZ}} = a_2 \, a_3 \, a_4 \, a_5 \ldots. \tag{3}$$

In this way the fatigue life can be calculated with the equation in ISO 281:1990

$$L_{n\text{a}} = a_1 \, a_2 \, a_3 \, L_{10} \tag{4}$$

or more generally, with the factor a_{XYZ} expressed as

$$a_{\text{XYZ}} = a_2 \, a_3 \, a_{\text{m}} \tag{5}$$

The a_{m} factor includes environmental variables that are not accounted for in ISO 281:1990. To fulfil the demand on a systems approach, it is however, important to understand the fact that life-influencing factors are interdependent, and a change in operating conditions can influence all factors in equations (3) to (5).

Annex A
(informative)

Equation for calculating the life adjustment factor a_1 for reliability

The life adjustment factors, a_1, in Table 1 of this Amendment are calculated with the aid of equation (A.1)

$$a_1 = \left(\frac{\ln \frac{100}{S}}{\ln \frac{100}{90}} \right)^{\frac{1}{e}} \qquad \text{(A.1)}$$

Table 1 is calculated for different reliabilities (probabilities of survival) S in % and a constant value 1,5 of the Weibull exponent e.

The equation (A.1) also makes it possible to calculate the factor a_1 for other Weibull exponents.

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[1] LUNDBERG, G. and PALMGREN, A., *Dynamic Capacity of Rolling Bearings*, Acta Polytechnica, 7 (1947), Mechanical engineering Series, Vol. 1, No. 3.

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